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STRENGTH OF BEAMS

UNDER

TRANSVERSE LOADS.

BY

WILLIAM ALLAN, A.M., LL.D.,

LATE PRINCIPAL OF McDONOGH SCHOOL, MARYLAND.
AUTHOR OF "CHANCELLORSVILLE," "JACKSON'S VALLEY CAMPAIGN," "THE ARMY OF NORTHERN VIRGINIA IN 1862," ETC., ETC.

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PREFACE.

The usual discussion of stress in beams loaded transversely, involving as it does the calculus, is unintelligible to that large class of builders, and others, whose mathematical training has not extended beyond the elements of conic-sections. Yet, the most important cases may be explained without using the higher mathematics.

The aim of the following pages is to put into brief and convenient shape, without resorting to the higher mathematics, the discussion of the most important and common cases of horizontal beams under vertical loads. In beams of rectangular cross-section the formulæ given are exact under the assumptions generally made in regard to the laws of elasticity. In flanged beams the approximations made use of are those adopted in ordinary practice.

A graphic method of determining the *amount of resistance* is added, which is applicable to all cross sections, and which, if used with ordinary skill and care, will give results exact enough for every purpose. This method is so simple, and of such general application, that it commends itself to

practical builders. It is easily understood, and requires no other appliances but those in the hands of every draughtsman. In the case of unsymmetrical cross-sections, (as for instance, iron rails) it affords an easy means of obtaining sufficiently accurate results, without resorting to the tedious calculations otherwise necessary.

This discussion (in which there is no attempt at originality) was prepared for the use of the intermediate classes in Engineering, at Washington and Lee University. Having been found serviceable to them, both during their college career and in subsequent practice, I am induced to hope that it may save time and labor to those engaged in the same pursuits.

In preparing the manuscript for the press, and correcting the proofs, I have been much indebted to one of my former pupils, Mr. Julius Kruttschnitt, C. E., now Instructor in the McDonogh School.

August, 1875.

STRENGTH OF BEAMS UNDER TRANSVERSE LOADS.

In the following discussion the ordinary cases of loaded beams are treated without resorting to the higher mathematics. It is an attempt to compile from various sources the simplest methods of treating such cases as arise most frequently in practice.

Transverse stress is produced by a load applied to a beam in a direction perpendicular, or inclined, to its length. A D (Fig. 1) is a beam subjected to such a stress.

In this kind of stress a compression of the particles or fibres on one side of the

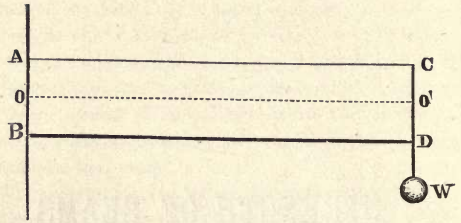


FIG. 1.

beam and an extension of those on the other, are produced. In consequence of this the beam *bends*. Experiment shows that the amount of compression on the one side, and of extension on the other, diminishes as we go inwards from the top or bottom towards the centre, and at some intermediate plane, O O′, becomes zero. The fibres at this plane being neither lengthened nor shortened, it is called the *neutral* plane, and its intersection by the plane of vertical section is called the *neutral line or axis*. Experiment also shows that, from this plane towards the top and bottom, the amount of extension and compression may, for the stresses that occur in ordinary prac-

tice, be considered as varying directly with the distance from the neutral plane.

The extreme top and bottom fibres suffer the greatest compression and extension, and in case of rupture, the rupture begins with them. Some question exists as to the exact location of the neutral line or plane, but for slight deflections it passes through the centre of gravity of the cross section of the beam, and it is very probable that it never deviates from this position.

In discussing transverse stress, the assumptions based upon experiment may be stated as follows:

1. The forces on the fibres are directly as the amount of extension or compression they produce; (*Ut tensio sic vis,*) and since the extension and compression increase as the distance from the neutral axis, the forces vary in the same proportion.

2. Within *elastic limits* the extension and compression at equal distances from

the neutral axis are equal, and the forces producing them are equal.

3. The *neutral axis* passes through the centre of gravity of the cross section.

RECTANGULAR BEAMS.

Let us now discuss the relations existing between the forces in, and on, transversely loaded rectangular beams, the load being supposed to be vertical in direction and the beam horizontal.

Case I.

Let A D (Fig. 2) be a beam so thin that it may be considered as composed of but one layer of fibres or particles. Let it be fastened in a wall at A B, and be loaded with W′ at the other end. Neglect for the time the weight of the thin beam itself, which is small. Imagine it to be cut by a vertical plane E F′ at any point, and let us see under what forces the part E D is held in equilibrium.

The only external force on E D is W′ acting at D downwards, and E D is pre-

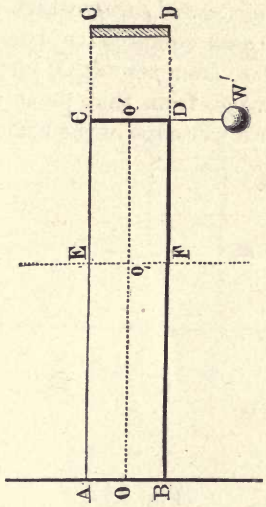
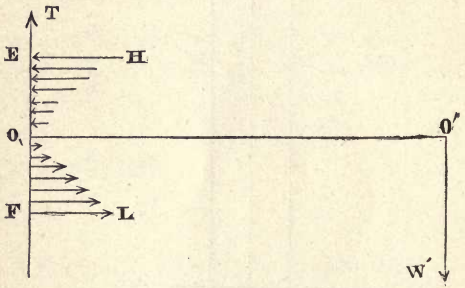


FIG. 2.

vented from falling under this weight by the *resistance of the fibres* at E F. To analize these forces, let us take O_1 as an origin of co-ordinates, and $O_1 O'$ as the axis of x, and O_1 E as the axis of y, and as the forces are all in one plane, find their components along these axes. The *internal* forces, or resistance of the fibres at E F, are:

1. The *horizontal forces* which are tensile above and compressive below, and which increase from zero at O_1 just in proportion as we go from that point towards the upper or lower edge of the beam. (Fig. 3.)



(Fig. 3.)

2. The *vertical force.* This is called the *shearing force* or *transverse* shearing force. It resists the tendency of the part of the beam E D to slide down on the surface E F. The existence of this force may be realized if we conceive the beam to be divided into two parts by the vertical plane E F, and those parts to be united by some very elastic substance, as india-rubber.

Then the beam would take the form shown in Fig. 4, the part F C sliding down on the

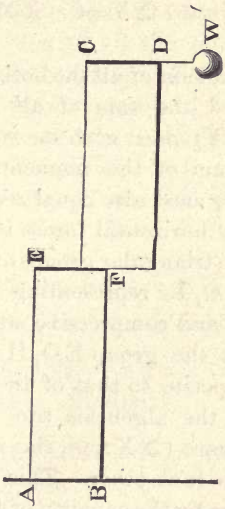


FIG. 4.

other. The force in the beam that resists this sliding is represented in Fig. 3, by the vertical arrow at E. Let it be called T. In Fig. 3 are represented all the forces we have to deal with. Since this system of

forces is balanced, the following equations must be fulfilled:

$$\Sigma X=0 \quad \Sigma Y=0 \quad \Sigma M=0 \qquad (1)$$

That is: the sum of all the horizontal forces (ΣX), and the sum of all the vertical forces (ΣY) must each be equal to zero, and the sum of the moments about any point as O_1 must also equal zero.

The only horizontal forces in the system are the two triangular groups of forces E O_1 H and F. O_1 L, representing the sum of the tensile and compressive stresses on the fibres. As the group E O_1 H acts in a direction opposite to that of the group F O_1 L, and as the algebraic sum of the two groups is zero ($\Sigma X=0$), the groups must be equal to each other. This is indicated in the figure by the equality of the triangles E O_1 H and F O_1 L.

The vertical forces are W′ and the shearing force T at E F, and since

$$\Sigma Y=T-W'=0. \qquad \text{We have}$$
$$T=W' \qquad (2)$$

Next obtain the moments of all the forces about O_1 and place the sum of these moments = zero. Replace the tensile and compressive forces by their resultants. The resultant or sum of all the tensile forces represented by the triangle E O_1 H (Fig. 3) may evidently be represented by the area of the triangle of which the base E O_1 is the distance over which the forces are distributed, and the altitude E H is the stress in the outside fibre. Let

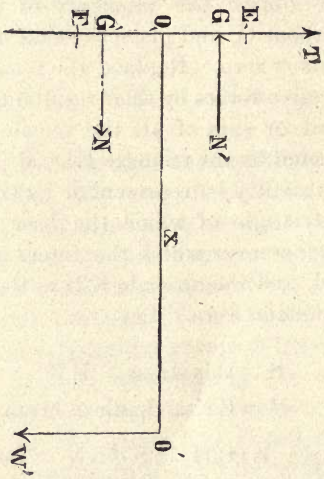
$$S = \text{this stress} = E\,H$$

and $$d = E\,F = \text{depth of beam}$$

Then area E O_1 H $= \frac{1}{4}\,S\,d = N =$ resultant of tensile forces. Similarly

Area F O_1 L $= \frac{1}{4}\,S\,d = N' =$ resultant of compressive forces.

These resultants will pass through the centres of gravity of the triangles E O_1 H and O_1 F L, since the little forces of which they are composed are represented by these triangles. Hence the direction of N will intersect O_1E at a point G (Fig. 5), whose



distance from O_1 is $= {}^2/_3 \, E O_1 = \frac{2}{3} \cdot \frac{d}{2} = \frac{d}{3}$.
This is the lever arm of N about O_1. That of N' is $G' O_1$ also $= \frac{d}{3}$. Hence the sum of the moments of these two forces about O_1 (since they both tend to produce left-handed rotation) is

$$-N \frac{d}{3} - N' \frac{d}{3} = -\frac{1}{6} S d^2.$$

The force T since its direction passes through O_1 has no lever-arm, and hence its moment is zero.

If the distance from O′ to O_1 be called x, the moment of the weight W′ is $= +W'x$ (since it tends to produce right-handed rotation).

$$\therefore W'x - \frac{1}{6} S d^2 = \Sigma M = 0$$

$$\therefore W'x = \frac{1}{6} S d^2 \qquad (3)$$

So far we have considered a beam whose breadth is that of only one row of fibres, but a beam of any breadth may be made up of a number of such slices placed side by side, and if b = the number of slices, or breadth of the beam, and W = the weight hung at the end of it, then eq. (3) becomes

$$W x = \frac{1}{6} S b d^2 \qquad (4)$$

This discussion is general and will apply to any section as well as to EF. S and x are the variables in eq. (4), and these quantities will have different values at the different sections, which values in-

crease as we go towards A B (Fig. 2), but the form of the equation will evidently be unchanged. If A C (Fig. 2) be $= l$, we have for the section at A B (Fig. 2)

$$W l = \frac{1}{6} S b d^2 \qquad (5)$$

A B is the section of greatest stress, and the beam if overloaded will break there.

The quantity $\frac{1}{6} S b d^2$ is called the *moment of resistance* of the fibres, or moment of the internal forces, and is often written M for brevity. Wx is called the moment of the weight, or moment of the external forces.

Let the maximum value of $\frac{1}{6} S b d^2$ (eq. 5) be called M_0.

We may illustrate geometrically the variation of the moments $M = Wx$, and consequently of the stresses produced on the outside fibres from A to C.

In Fig. 6 let A C be the beam. Take a line on some scale to represent the value

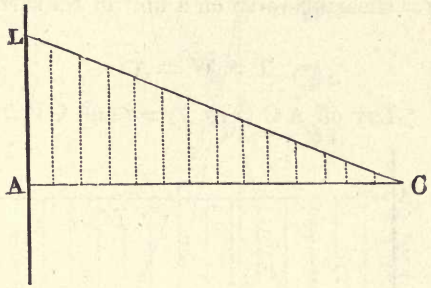


FIG. 6.

of $M_0 = W l$, and lay it off from A perpendicular to A C. Let A L be this line. Draw L C. Then the dotted perpendiculars in the triangle L A C will represent the moments of resistance in the beam at the several points at which they are drawn.

From eq. (2) it is seen that the shearing force is constant at every section of the beam. This force we may assume with sufficient accuracy, for our present purpose, to be uniformly distributed over the cross section of the beam on which it acts. Hence if A = area of cross section and,

t = shearing force on a unit of the surface,

$$T = W = A\,t \qquad (6)$$

Lay off A C (Fig. 7) = l and C P = W.

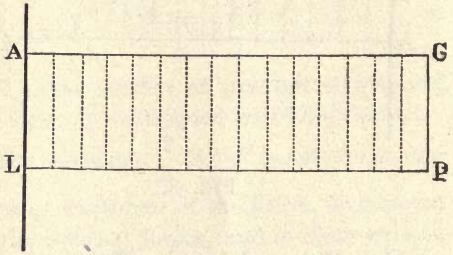


Fig. 7.

Then the rectangle A P represents geometrically the shearing stress at every point of the beam.

Corollary. When several weights as W W_1 W Fig. 8) are suspended from the beam at different points, the moment of resistance at any point is equal to the joint moments of the weights at that point. Thus, calling distances measured from C, G, and E towards A, x, x_1 and x_2 respect-

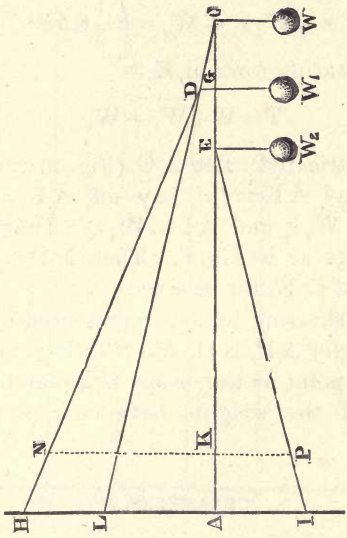


FIG. 8.

ively, we have for the equation of moments for points between C and G

$$W x = \frac{1}{6} S b d^2 \qquad \text{Between G and E}$$

$$W x + W_1 x_1 = \frac{1}{6} S b d^2 \qquad \text{While at K,}$$

for instance, it is

$$W x + W_1 x_1 + W_2 x_2 = \frac{1}{6} S b d^2 \qquad (7)$$

The shearing force at K is

$$T = W + W_1 + W_2 \qquad (8)$$

Geometrically. Let A C (Fig. 8) $= l$ A G $= l_1$ and A E $= l_2$. Lay off A L $= W l$, L H $= W_1 l_1$ and A I $= W_2 l_2$. Draw the triangles as in Fig. 8. Then N P = total moment at K, for instance.

The shearing force is represented by the rectangles A P, N O, and S T (Fig. 9), and at any point in the beam is equal to the sum of the weights between that point and C.

EXAMPLES.

(1.) Suppose the safe stress per square inch to be 1,000 lbs. (= S), and $l = 10$ ft., $b = 3$ inches, and $d = 12$ inches, what weight will the beam support?

(2.) Suppose W = $^3/_4$ ton, $l = 12$ ft., $b = 2$ inches, what must be the depth (d) of a rectangular cast iron beam, so that S shall not exceed 4 tons?

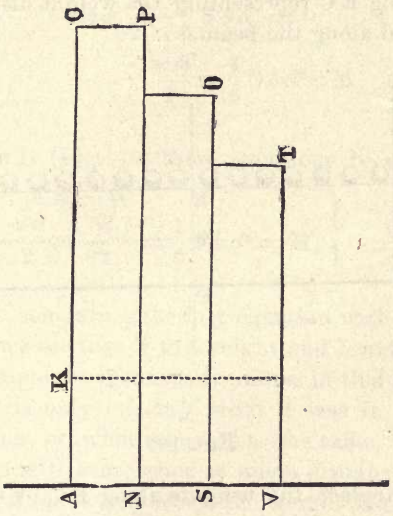


FIG. 9.

Case II.

Let the beam be as in the last case, but with the load distributed uniformly over it (Fig 10). Let w = weight on a unit of length, W = total weight on A C, l = A C = length, d = A B = depth, x = E C as before. Then the forces to be considered are represented in Fig. 11, the little arrows

along E C representing the weight distributed along the beam.

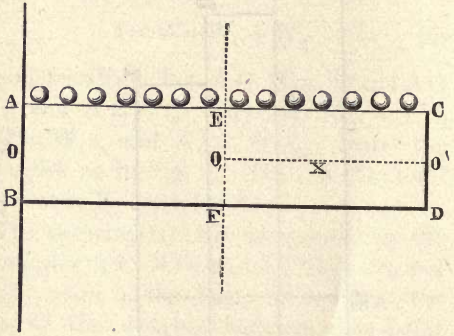


FIG. 10.

Replace the weights along E C by their resultant, which is $= w\,x$, and which should be applied at the middle point of E C, since the little weights on the beam are uniformly distributed. Then putting the resultants N and N′ in place of the tensile and compressive force, and proceeding as before, we have

$$T - w\,x = 0$$
$$T = w\,x \qquad (9)$$

Also $$w\,x.\frac{x}{2}-\frac{1}{6}\,\mathrm{S}\,b\,d^2=0$$

$$\frac{w\,x^2}{2}=\frac{1}{6}\mathrm{S}\,b\,d^2=\mathrm{M} \qquad (10)$$

At A B (Fig. 10) these equations become

$$\left.\begin{aligned} &\mathrm{T}_0=wl=\mathrm{W}\\ &\frac{w\,l^2}{2}=\frac{\mathrm{W}\,l}{2}=\frac{1}{6}\,\mathrm{S}\,b\,d^2=\mathrm{M}_0 \end{aligned}\right\} \qquad (11)$$

By comparing the last equation with eq. (5), we see that if the weight and beam be the same the stress on the fibres in this last case is only *one-half* what it was in the former, or, what amounts to the same, the beam will bear twice as much distributed over it, as it will when the weight is concentrated at the extremity.

From eq. (9) we see that the shearing force is not constant as in the last case, but varies as x. It is greatest at A.

Geometrically. The equation $\mathrm{M}=\frac{1}{2}\,w\,x^2$ corresponds with that of a parabola with vertex at C and axis vertical. Lay off A L (Fig. 12) $=\frac{1}{2}\,w\,l^2$, and through L and C

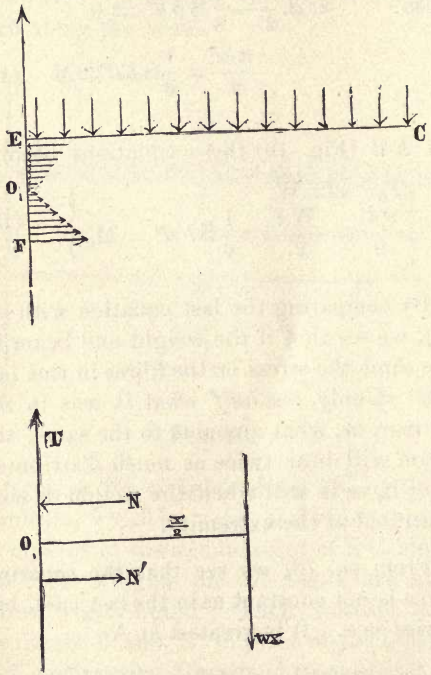


FIG. 11.

draw a parabola. The ordinates of this parabola (dotted in the figure) will represent the moments at the several points.

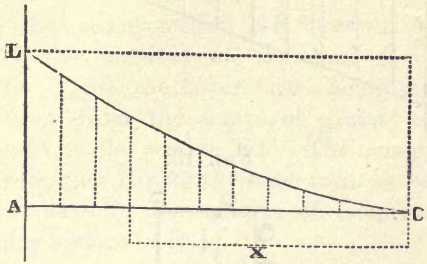


FIG. 12.

The equation $T = w\,x$ is represented by the triangle A P C (Fig. 13), which therefore gives the shearing stress at every point of the beam when A P is taken $= w\,l$.

Corollary 1. When the load is distributed over only a part of the beam as in Fig. (14), let R C$=m=$the loaded part, and take the other letters as before. Then the equations for any section in the loaded part are evidently the same as those just obtained, viz.:

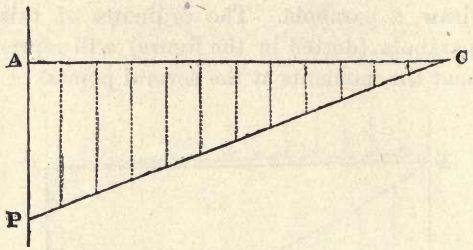


FIG. 13.

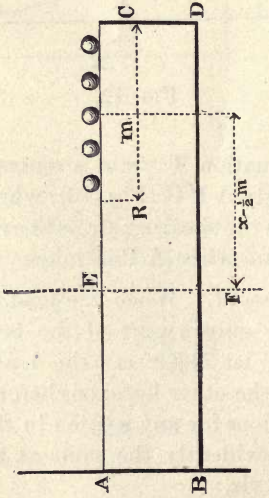


FIG. 14.

$$\left.\begin{array}{rc} & {}^1/_2\, w x^2 = \frac{1}{6} S b d^2 = M \\ \text{At R} & {}^1/_2\, w m^2 = M \\ \text{And} & T = w x \end{array}\right\} \quad (12)$$

But at any section E F between A and R the moment of the load is $= w m (x - {}^1/_2 m)$, the latter factor being the distance from the centre of gravity of the load to the section E F. The moment of resistance having the same form as before, we have for the equation of moments for any section in R A

$$w m (x - {}^1/_2 m) = \frac{1}{6} S b d^2 = M \quad (13)$$

At A this becomes
$w m (l - \frac{1}{2} m) = M_0$ the greatest moment.

The shearing force at E F being equal in amount and opposite in direction to the whole load between E F and C will be

$$T = w m \quad (14)$$

Geometrically. For the moments: lay off A C $= l$ and C R $= m$ (Fig. 15). At R erect D R $= \frac{w m^2}{2}$ and at A make A L $= M_0$.

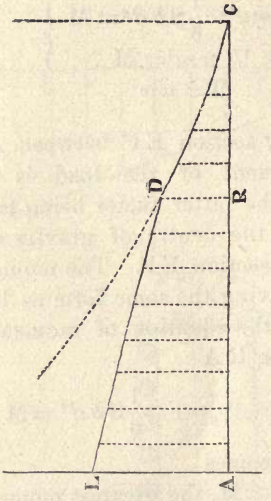


Fig. 15.

Through C and D draw a parabola as in the last case, and (since eq. [13] is of the first degree) through D and L draw a straight line. Then from C to R the ordinates of the parabola represent the moments, and from R to A they are represented by the ordinates of the trapezoid R L. For the shearing stress (Fig. 16), lay off from

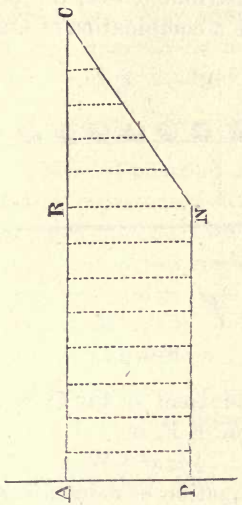


Fig. 16.

R, R N$=w\,m$. Draw C N and N P. Then the triangle C R N (corresponding to the equation T$=w\,x$) gives the shearing force at each point in C R while the rectangle R P (corresponding to the equation T$=w\,m$) gives the force in the remaining segment of the beam.

Corollary 2. When there is a load W at the extremity C, in addition to the load

uniformly distributed over the beam (Fig. 17), we have a combination of Cases I. and

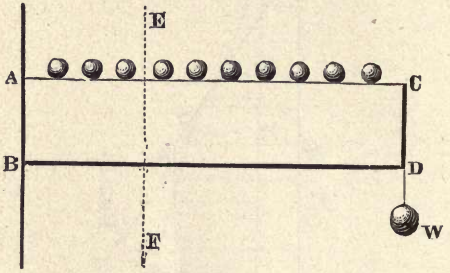


FIG. 17.

II. and the moment of the external forces at any section, E F, is

$$\tfrac{1}{2} w x^2 + W x$$

Hence the equation of moments is

$$M = \frac{1}{6} S b d^2 = \tfrac{1}{2} w x^2 + W x \qquad (15)$$

This is greatest at A, or

$$M_0 = \frac{1}{6} S b d = \tfrac{1}{2} w l^2 + W l \qquad (16)$$

The shearing force

$$T = w x + W \qquad (17)$$

At A

$$T_0 = w l + W \qquad (18$$

Geometrically. The simplest way of representing the moments is to construct those due to each kind of weight, and then combine them. Thus, let $M' = \frac{w x^2}{2}$ and $M'' = W x$. Construct M′ as in *Case II.*, it being represented by a parabola with vertex at C and axis vertical, and M″ as in *Case I.*, it being represented by a triangle (placed under A C for convenience). Since $M = M' + M''$ from eq. (15) we have the total moment at any point E (Fig. 18) represented by the sum of the ordinates of these two figures = N P Fig. (18.)

The moments may also be represented by the parabola corresponding to eq. (15) as in Fig. (19.) This parabola has its vertex at C′ and not at C. Of course, only that part of the curve between C and A is applicable to our purpose.

The shearing force is represented by adding the triangle N P P′ (Fig. 20) representing the variable part $w x$ of T to the rectangle C P which represents the constant part W of T.

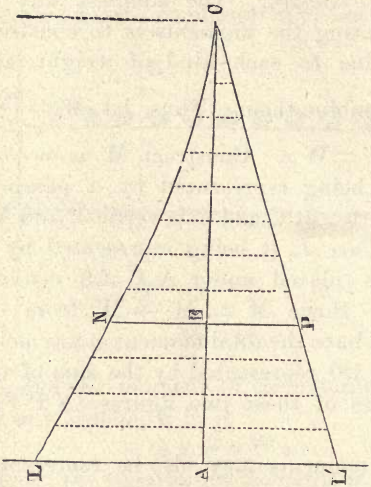


Fig. 18.

EXAMPLE.

Discuss the forces when the load is distributed as shown in Fig. (21) assuming various values for L N and N C as well as for w and W.

Case III.

Let the beam whose length is l rest upon

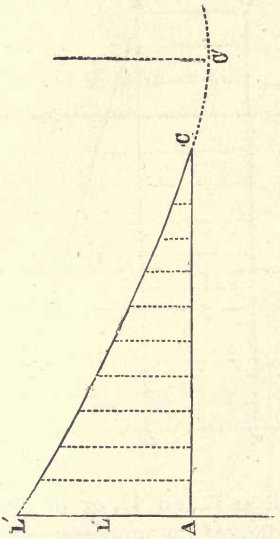


Fig. 19.

supports at B and D (Fig. 22) and let it be loaded at some point G with a single weight W. Let m and n be the segments into which the beam is divided at the point G of the application of the weight.

First find the proportions of the weight

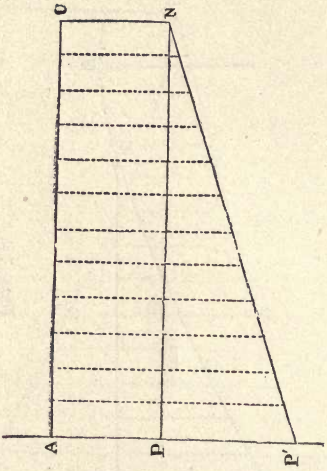


FIG. 20.

supported at B and D, or in other words, the reactions of the supports. By the principle of the lever the respective portions of the weight supported at B and D are inversely proportional to the distances of these points from G. Thus, let W′=reaction at D and W″=reaction at B and then

$$W' : W'' : : m : n$$
$$W''+W' : W' : : m+n : m$$

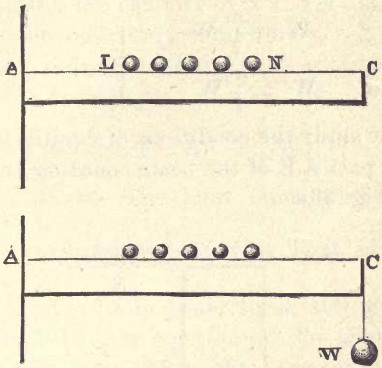


FIG. 21.

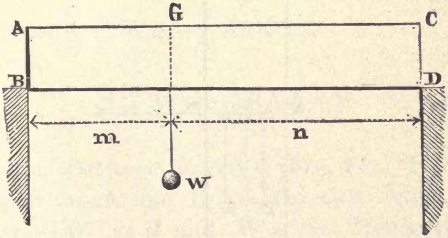


FIG. 22.

But $W' + W'' = W$ and $m + n = l$

$$\left.\begin{aligned} \therefore W' &= \frac{m}{l} W \\ \text{And so} \quad W'' &= \frac{n}{l} W \end{aligned}\right\} \qquad (19)$$

Now apply the conditions of equilibrium to any part A E of the beam counting from A. (Fig. 23).

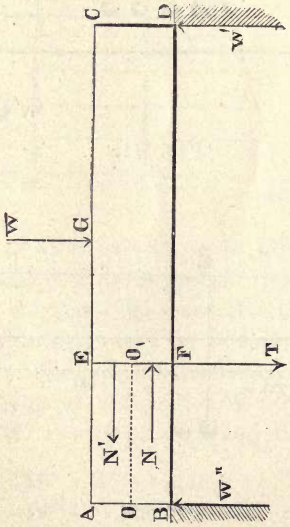


FIG. 23.

1st. Between A and G. $\Sigma X=0$ merely indicates the equality of N and N′, as these are the only horizontal forces. $\Sigma Y=0$ shows that the shearing force at the section E F is downwards

$$\text{and}=W'' \quad \therefore T=W''=\frac{n}{l}W \qquad (20)$$

$\Sigma M=0$. The joint moment of N and N′ is as before $=\frac{1}{6}S\,b\,d^2$. That of T is zero. The only other force acting on A E is W″, the reaction of the abutment. Let $O\,O_1=x$. Then the moment of W″ is

$$W''x=\frac{n}{l}.W\,x$$

Hence $$\frac{n}{l}.W\,x-\frac{1}{6}S\,b\,d^2=0$$

$$\therefore \frac{n}{l}\,W\,x=\frac{1}{6}S\,b\,d^2=M \qquad (21)$$

2d. Between G and C (Fig. 24). Here, between A and E are the two external forces W″ at B and W at G. Hence the shearing force at E F is upwards and

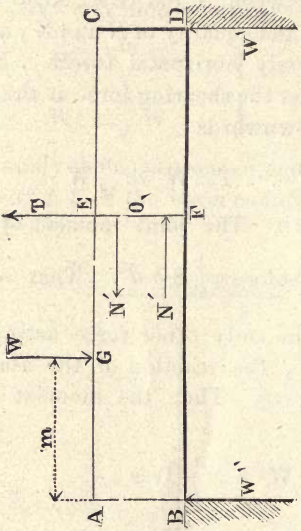


FIG. 24.

$$T = W'' - W = -W' = -\frac{m}{l}W. \qquad (22)$$

For $\Sigma M = 0$ we have

$$-\frac{1}{6}S\,b\,d^2 + W'x - W(x-m) = 0$$

$$\therefore \frac{n}{l}Wx - W(x-m) = \frac{1}{6}S\,b\,d^2 = M \qquad (23)$$

The greatest value of eq. (21) is at G, where it becomes identical with eq. (23) for the same point. This value is

$$\frac{m.\,n}{l}\mathrm{W}=\frac{1}{6}\mathrm{S}b\,d^2=\mathrm{M}_0 \qquad (24)$$

At A and C the moments are zero.

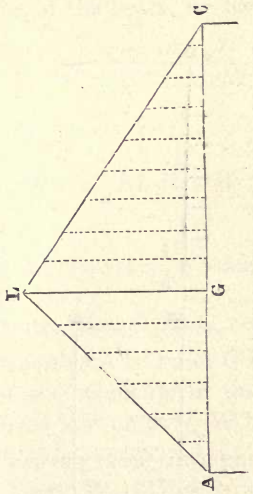


FIG. 25.

Geometrically. From eq. (21), which is of the first degree, it is seen that the moments

vary in A G as they did in *Case I.* Hence they may be represented by the ordinates of the triangle A L G. (Fig. 25).

Eq. (23) is also that of a straight line cutting the axis of X at C. Hence the moments in G C are represented by the triangle G L C (Fig. 25). The shearing

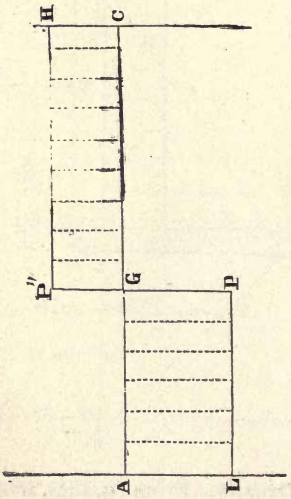


FIG. 26.

force in A G is represented by the rectangle A P and that in G C by the rectangle CP'. (Fig. 26.)

Note. That the maximum moment (at G) corresponds to the point where the shearing force passes through zero.

Corollary 1. When the weight W is at the middle of the beam, we have

$$n = m = {}^1/_2\, l$$
$$W' = W'' = {}^1/_2\, W$$

Then eq. (21) becomes

$$\left.\begin{aligned} {}^1/_2\, W x &= \frac{1}{6} S b d^2 = M \\ \text{and (23) is} \quad {}^1/_2\, W (l - x) &= \frac{1}{6} S b d^2 = M \end{aligned}\right\} \quad (25)$$

At the centre $M_0 = {}^1/_2 W . {}^1/_2\, l = {}^1/_4 W l$ (26)

The triangles A L G and G L C (Fig. 27) represent the moments in this case. G L having been laid off $= {}^1/_4\, W\, l$.

The shearing force throughout the beam is then $T = {}^1/_2 W$ (27) as is shown in the rectangles (Fig. 28).

Comparing the value of M_0 given in eq.

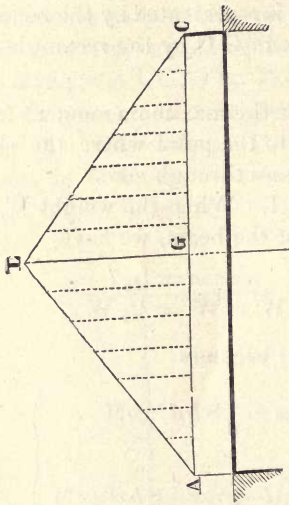


Fig. 27.

(5) *Case I.* with that of M_0 in eq. (26) we see that the load and the length being the same, a beam will bear *four* times as much with both ends supported, and the load placed in the middle as it will do with one end fixed and the other loaded.

Corollary 2. When there are several weights, as in Fig (29), the moment of the

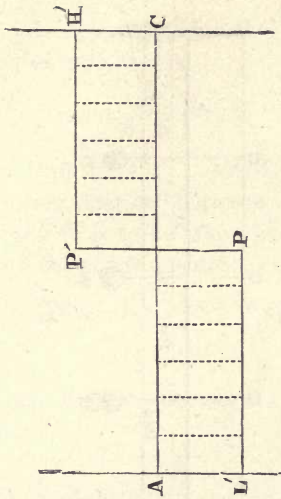


Fig. 28.

external forces at any section is that due to the action of all the weights. Let the segments into which the weights W, W_1 and W_2 divide the beam be m and n for W, m_1 and n_1 for W_1 and m_2 and n_2 for W_2.

Let R_1 = reaction of abutment at B and
R_2 = reaction of abutment at D and
l = length and let x be counted from A as before.

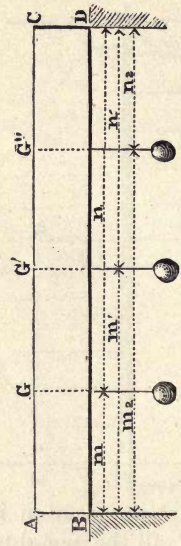


FIG. 29.

The reaction of the abutment at B due to these weights is

$$\left.\begin{aligned} R_1 &= \frac{n}{l}W + \frac{n_1}{l}W_1 + \frac{n_2}{l}W_2 \\ \text{So at D } R_2 &= \frac{m}{l}W + \frac{m_1}{l}W_1 + \frac{m_2}{l}W_2 \end{aligned}\right\} \quad (28)$$

For any section between A and G, R_1

is the only external force and hence the equation of moments is

$$R_1 x = \frac{1}{6} S\, b\, d^2 = M \qquad (29)$$

The shearing force is $T = R_1$ (30)

For every section between G and G′ the weight W is to be taken into consideration and the equations are:

$$R_1 x - W(x-m) = \frac{1}{6} S\, b\, d^2 = M \qquad (31)$$

$$T = R_1 - W \qquad (32)$$

For any section between G′ and G″) we have:

(33)

$$R_1 x - W(x-m) - W_1(x-m_1) = \frac{1}{6} S\, b d^2 = M$$

$$T = R_1 - W - W_1$$

For any section between G″ and C

$$\left.\begin{aligned} R_1 x - W(x-m) - W_1(x-m_1) - W_2 \\ (x-m_2) = \frac{1}{6} S\, b\, d^2 = M \\ T = R_1 - W - W_1 - W_2 \end{aligned}\right\} \qquad (34)$$

The location of the greatest moment is most readily determined by geometrical construction.

Geometrically. The moments are represented in Fig. (30) by constructing sep-

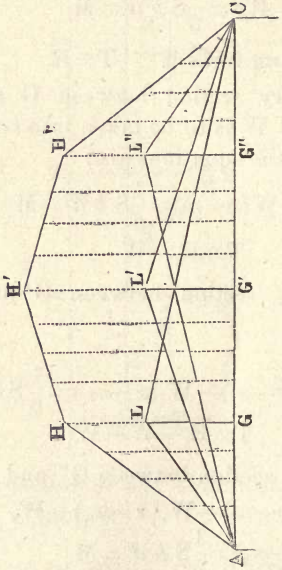


FIG. 30.

arately those due to each weight and then combining them. Thus, the moments produced at every point in the beam by the weight W are represented by the triangle

A L C (Fig. 30), in which G L equals the greatest moment due to $W=\frac{n}{l} W m$. Similarly A L′ C represents those due to W_1, G′ L′ being $=\frac{n}{l}W_1 m_1$ and A L″ C gives those due to W_2, G″ L″ being $=\frac{n_2}{l} W_2 m_2$.

Now, if at every point we add together the ordinates of these three triangles for that point, and lay them off above A C we shall get a polygon A H H′ H″ C, which represents eqs. (29) (31) (33) and (34) and gives the total moment at any section. The greatest ordinate of this polygon will, of course, show the location of the maximum moment. This will be at G or G′ or G″, according to the relative amounts and positions of the weights W, W_1, and W_2. In the Fig. it is at G′. Hence from eq. (31)

$$R_1 m_1 - W(m_1 - m) = \frac{1}{6} S b d^2 = M_0 \quad (35)$$

The shearing force may be represented

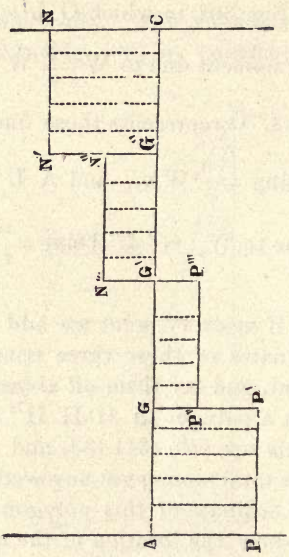


FIG. 31.

as in Fig. (31). It is greatest in that one of the two end segments which corresponds to the greater of the two quantities R_1 and R_2.

Note. That in this case the simplest way of finding the point of maximum moment

is to construct the figure representing the shearing force, and the point when the shearing force passes through zero, (G′ Fig. 31) is the point sought.

EXAMPLES.

1. Let l=20 ft., m=5 ft., m_1=10 ft., m_2=15 ft., W=1 ton, W_1=2 tons, W_2=3 tons. Find the maximum moment.

2. Find the size of a rectangular wooden beam where

l=15 ft., m=3 ft., m_1=6 ft., m_2=14 ft.
W=1 ton, W_1=½ ton, W_2=2 tons
S=1,000 lbs. and d=4 b.

Case IV.

Let the load be equally distributed over the beam (Fig. 32). In this case the reaction of each abutment=½ the load, or

$$R_1=\frac{w\,l}{2}=R_2 \qquad (36)$$

Take any section E F whose distance from A=x. Then the external forces

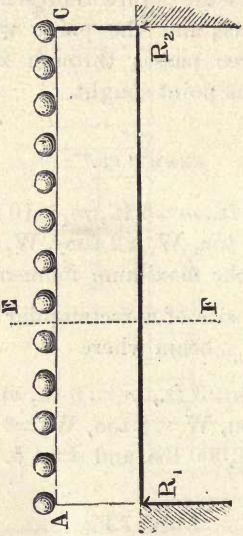


FIG. 32.

acting between A and E F are, R_1 and the resultant of all the little weights from A to E ($=w\,x$). This last force acts at its centre of gravity (Fig. 33), which is half way from A to E. Its lever arm is therefore $=\frac{x}{2}$. Hence the equation of moments will be

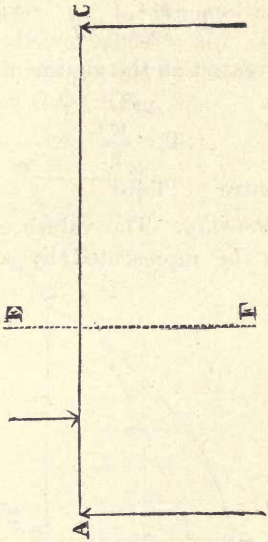


FIG. 33.

$$\left.\begin{aligned} &\frac{wl}{2}x - \frac{wx^2}{2} = \frac{1}{6}\mathrm{S}\,b\,d^2 = \mathrm{M} \\ \text{or}\quad &\frac{wx}{2}(l-x) = \frac{1}{6}\mathrm{S}\,b\,d^2 = \mathrm{M} \end{aligned}\right\} \qquad (37)$$

This is a maximum at the centre where

$$\mathrm{M}_0 = {}^1/_8\, w\, l^2 \qquad (38)$$

The shearing force at E F is

$$T = \frac{w\,l}{2} - w\,x \qquad (39)$$

This is greatest at the abutments where $x = l$, or 0

$$\therefore T_0 = \frac{w\,l}{2} \qquad (40)$$

At the centre $T = 0$

Geometrically. The values of M in eq. (37) may be represented by a parabola

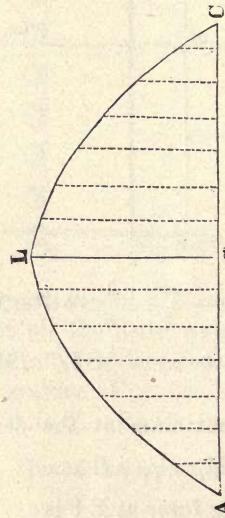


FIG. 34.

with vertex at L, the ordinates G L (Fig. 34) being taken $= M_0$. The shearing force is represented by the two triangles A P G and G Q C (Fig. 35).

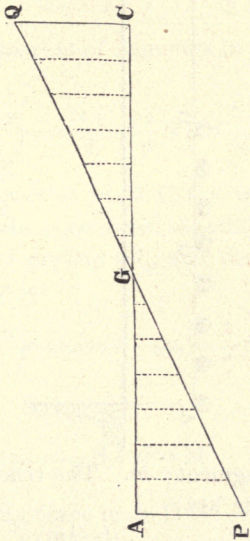


Fig. 35.

The maximum moment exists at the point (G) when the shearing force is zero.

Corollary. When the uniform load extends only over a certain distance from one of the supports, as in (Fig. 36), let A D=

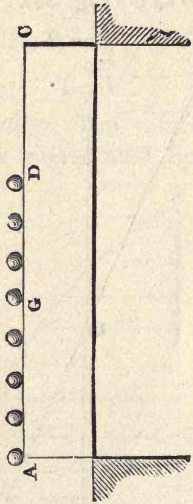


Fig. 36.

loaded segment=m. The reactions of the abutments are:

$$\left.\begin{aligned} &\text{At A, } R_1 = w\,m\left(\frac{l-\frac{1}{2}m}{l}\right) \\ &\text{At C, } R_2 = w\,m\left(\frac{m}{2\,l}\right) \end{aligned}\right\} \qquad (41)$$

Then for any section in A D the moments of the external forces will be as in the case just discussed,

$$= w\ m\ \left(\frac{l-\frac{1}{2}\ m}{l}\right) x - \frac{w\ x^2}{2}$$

And the equation of moments in A D will be:

$$w\ m\left(\frac{l-\frac{1}{2}\ m}{l}\right) - x\frac{w\ x^2}{2} = \frac{1}{6}\ \mathrm{S}\ b\ d^2 = \mathrm{M} \qquad (42)$$

For any section in D C the whole load ($w\ m$) is to be considered as acting through its centre of gravity (G), and the equation of moments is:

$$w\ m\left(\frac{l-\frac{1}{2}\ m}{l}\right) x - wm(x-\tfrac{1}{2}m) = \frac{1}{6}\ \mathrm{S}\ b\ d^2 = \mathrm{M}$$

Reducing

$$\frac{w\ m^2}{2\ l}(l-x) = \frac{1}{6}\ \mathrm{S}\ b\ d^2 = \mathrm{M} \qquad (43)$$

The shearing force in A D is

$$\left.\begin{array}{l} \mathrm{T} - w\ m.\left(\dfrac{l-\frac{1}{2}\ m}{l}\right) - w\ x \\ \text{In D C, } \mathrm{T} = w\ m.\left(\dfrac{l-\frac{1}{2}\ m}{l}\right) - w\ m \end{array}\right\} \qquad (44)$$

T is a maximum at A, or

$$T_0 = w\, m\left(\frac{l - \frac{1}{2}\, m}{l}\right)$$

Geometrically. Eq. (42) corresponds to the parabola A L K (Fig. 37), which cuts

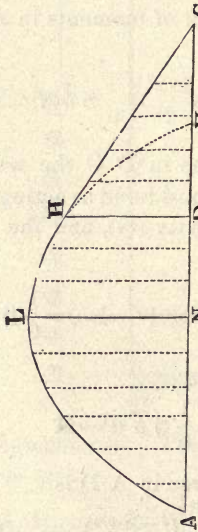


Fig. 37.

A C at A and K (whose distance from A = $\frac{2\, m}{l}\,[l - \frac{1}{2}\, m]$) and whose axis is vertical

Eq. (43) corresponds to the straight line H C. We can only use the part A L H of the parabola, the moments in D C being represented by the triangle D H C. The maximum moment is at N corresponding to the vortex L of the parabola. The value of this moment is:

$$M_0 = \frac{w\, m^2}{2}\left(\frac{l - \frac{1}{2}\, m^2}{l}\right) \qquad (45)$$

which is obtained from eq. (42) by substituting for x the value A N ($=\frac{1}{2}$ A K)$=$ $\frac{m}{l}(l - \frac{1}{2}\, m)$. This M_0 is always less than the maximum moment that exists when the load extends all over the beam as will appear by making m to vary in eq. (45) and applying the tests for a maximum to it.

The shearing stress for the loaded segment is represented by the triangles A P K and K P′ D (Fig. 38), and for the other segment by the rectangle D H. The point K, where the stress is zero, is found by making in eq. (44)

$$T = w\,m\left(\frac{l-\frac{1}{2}m}{l}\right) - w\,x = 0$$

and finding the value of x.

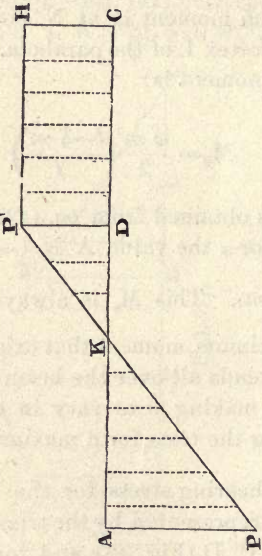


Fig. 38.

This point corresponds to the maximum moment. It may also be found graphically by constructing Fig. 38, and, as before, affords the easiest method of deter-

mining the point of the beam where the maximum moment exists and where consequently there is greatest danger of rupture.

EXAMPLES.

1. Let l=20 ft. w=500 lbs. per ft. m=15 ft. and let there be a weight in addition.

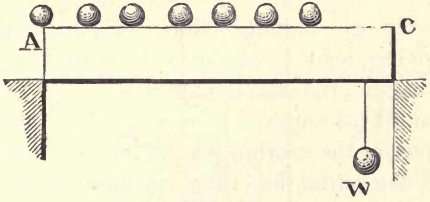


FIG. 39.

W=5 tons at a point 18 ft. distant from A. Required the maximum moment.

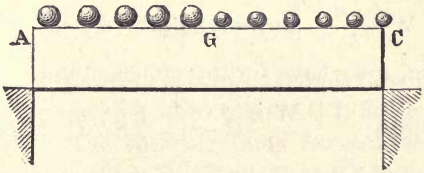


FIG. 40.

2. Let one-half of the above beam be loaded with a uniform load, $w=1$ ton per foot, and the other half with a uniform load of $w' = \frac{1}{2}$ ton per foot. Required the moments.

Case V.

A single moving load. When a single moving load passes over a beam, as in (Fig. 41), the maximum moment at each instant (as appears from *Case III.*) takes effect at the section just under the weight. To determine the law of variation of these maxima as the weight travels over the beam: Let x=the distance at any instant from A, and then the reaction of A at that instant (= the part of W transmitted to it) $= W\frac{l-x}{l}$ Multiplying this by the lever arm x we have for the moment under the weight:

$$\frac{W\,x}{l}(l-x)=\frac{1}{6}S\,b\,d^2=M \quad (46)$$

This is a maximum at the centre, where

$$M_0 = {}^1/_4\, W l \qquad (47)$$

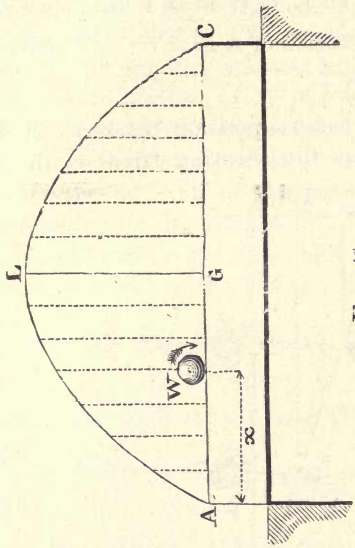


FIG. 41

Eq. (46) corresponds to a parabola (Fig 41) with vertex at L, the ordinate G L being $= {}^1/_4\, W l$. The shearing force for each segment into which W (Fig. 42) at any instant divides the beam is equal to the reaction of

the abutment corresponding to that segment. Thus, if W is at a distance x from A the reaction of A is $= W\frac{l-x}{l}$ and of C it is $= W\frac{x}{l}$.

If W has the position marked 2 in (Fig. 42), then the shearing stress in the left

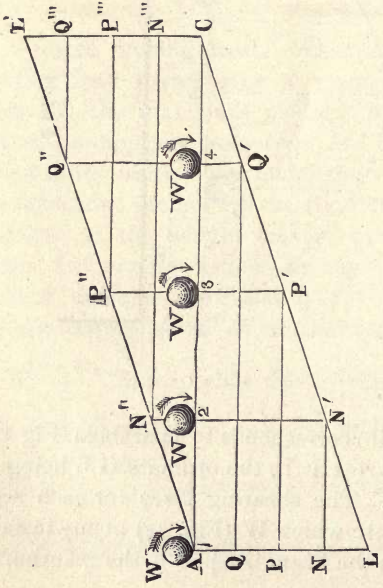


Fig. 42.

segment is shown by the rectangle A N N′ W and in the right segment by the rectangle W N″ N‴ C. The diagram shows in a similar way the stress at other points. If the third position of W in the figure is at the centre of the beam then evidently the greatest shearing stress to be provided for in the left half of the beam will be represented by the *locus* of the points like L, N′, P′, and for the right half it will be the *locus* of the points P″, Q″, L′, etc.

These *loci* are given by the equations :

$$\left.\begin{aligned} T &= \frac{W}{l}(l-x) = \text{eq. of L C} \\ T &= \frac{W}{l}x \quad\quad = \text{eq. of A L}' \end{aligned}\right\} \qquad (48)$$

Turn the triangle A L′ C down for convenience, as in (Fig. 43), and then the shearing stress to be provided for is given by the figure A L P L′ C. In this figure

$$A\,L = C\,L' = W \text{ and } D\,P = \frac{W}{2}.$$

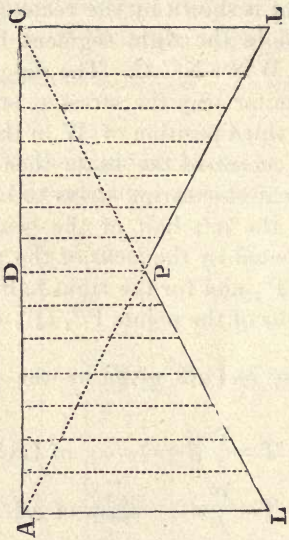


Fig. 43.

Case VI.

A distributed moving load. When a moving load gradually covers a beam, (Fig. 44), moving on from one end as a long train of cars, the maximum moments produced is that due to the load when it covers the entire length of the beam, and conse-

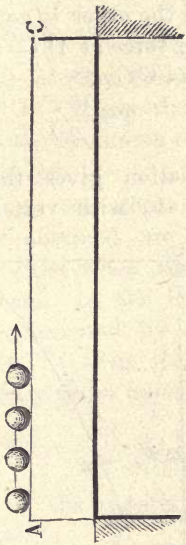


FIG. 44.

quently this case is provided for in *Case IV.*

But with the shearing stress it is different. Here, as in *Case V.*, we need the *locus* of the greatest shearing stresses that be brought upon the beam. This maximum at any section D occurs when the longer segment into which D divides the beam is

loaded, and the other is not. In that case the shearing force at D (=the reaction of the abutment C) is

$$T = \frac{w\,x^2}{2\,l} \qquad (49)$$

This equation gives the parabola A N P′ (Fig. 45) with vertex at A, where

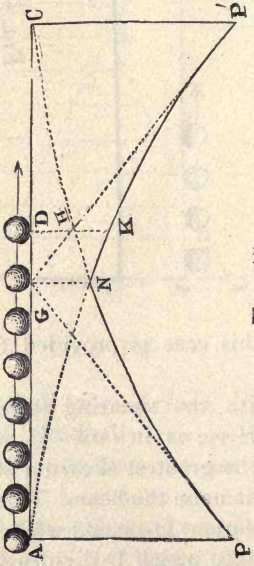


FIG. 45.

$C P' = \frac{w\,l}{2}$. When the load comes from the other end of the beam we get the parabola C N P. Hence the figure A P N P′ C gives the maximum shearing stress to be provided for.

It is easy to see that the shearing stresses thus obtained are greater than those which exist when the load covers the entire beam. In the latter case the forces are represented by the triangles A P G and G P′ C (Fig. 45), the shearing stress at D being given by eq. (39)

$$T = w\ x\frac{w\,l}{2} = D\ H$$

In the case of the passing load we have just seen that

$$T = \frac{w\,x^2}{2\,l} = D\ K$$

The value of D H is always less than that of D K when $x > \frac{l}{2}$; for if $2\,x$ be a certain quantity, then the product of the halves of that quantity ($= x^2$) is greater than the

product of any other two parts (such as l and $[2\,x-l]$) into which it can be separated.

That is
$$\left.\begin{array}{l} x^2 > l\,(2\,x-l) \\ \therefore \quad \dfrac{w\,x^2}{2\,l} > \dfrac{w\,l\,(2\,x-l)}{2\,l} \\ \text{or} \quad \dfrac{w\,x^2}{2\,l} > w\,x - \dfrac{w\,l}{2} \end{array}\right\} \qquad (50)$$

In the expressions of the moment of resistance $M=\frac{1}{6}\,S\,b\,d^2$ the quantity denoted by S (= the stress on the outside fibres) varies directly as M. Hence, all the geometrical illustrations we have given of the *moments* may apply equally well to the values of S. The maximum moments give the maximum stress on the fibres, and indicate the points of rupture when the beam is loaded with its breaking weight.

ULTIMATE VALUES OF S.

If beams are loaded transversely until fracture takes place, the value of S or the stress on the outside fibre which exists at

the moment of fracture, gives us a value for the tensile or compressive strength of the material according to the manner of rupture. If the beam yields by *tearing*, S gives us the tensile strength, if by *crushing* S gives the compressive strength. We readily obtain the value of S answering to the ultimate strength from any of the formulas under "Transverse Stress," by substituting given values for *l*, *b*, and *d*, and the actual breaking weight for W.

But the tensile and compressive strengths of materials are also obtained by direct tension and compression, the force being applied in the direction of the length of the bars until rupture takes place.

If our theory were perfect the values of tensile and compressive strength thus deduced would agree with the ultimate values of S found in tranverse stress; but they do not.

The difference is very wide sometimes. Thus in cast iron, S (in this case it represents the tensile strength) derived from

breaking rectangular beams by a transverse load is nearly 20 tons per square inch, while the tensile strength obtained directly is only about 8 tons. This discrepancy has been accounted for in two ways.

1. That the neutral axis moves towards the compressed side, and that therefore a larger portion of the beam is subjected to tension than the formula supposes.

2. That the neutral axis always remaining at the centre of gravity of the beam, the additional strength is due to the *adhesion* of the fibres which is developed by the unequal lengthening, and shortening of them as we go from the neutral axis towards the surfaces. In favor of this view is the fact that we know such adhesion to be an element of strength; for the compression or extension due to a given force is not so great in a transversely loaded beam as in one directly compressed or extended.

The action of this adhesive force may be illustrated as follows:

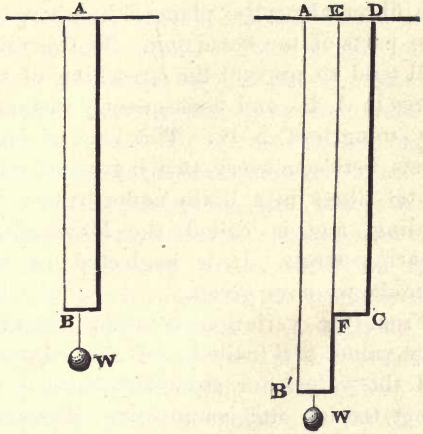


FIG. 46.

In the beam A B (Fig. 46), strained by the weight W, all the fibres are equally elongated, and they only resist by their direct tenacity. But in the beam A′C to the one-half of which is appended the weight, while the other half, E C, is less strained or altogether prevented from extending, evidently W will have to overcome not merely the tenacity of the fibres in A′ B′ but the adhesive force of

the fibres along the plane E F, where the two parts of the beam join; for this force will tend to prevent the stretching of the fibres in A′ B′, and consequently increases the strength of A′ B′. This kind of force exists between every two layers of horizontal fibres in a beam under transverse loading, and is called the *longitudinal* shearing stress. It is neglected in the formulæ we have given.

From the variation between the ultimate values of S (called *moduli of rupture*) and the values for strength obtained by direct tension and compression, it results that the values should be determined in both ways, and that the values gotten by one method should not be used in calculations involving the other kind of stress.

BEAMS OF UNIFORM STRENGTH.

As already stated, in solid rectangular beams, S has different values for the various points in the length of the beam. There is always a point of maximum

stress where the beam, if loaded sufficiently, will break. Now at all other points there is an excess of material which is useless and injurious from its own weight. To secure the requisite strength with the least material is an object usually desirable, and this can be readily accomplished in certain materials (as cast iron), by giving the beam such a shape as will make S, the stress on the outside fibre, constant throughout its length. In wood the injury resulting from the cross cutting of the fibres frequently prevents the putting of the theory into practice.

The application of the theory of uniform strength to beams of *rectangular cross section* may be most simply explained by taking up the cases we have discussed in detail.

In *Case I.* from eq. (5) the maximum stress in the outside fibre is,

$$S = \frac{6\,W\,l}{b\,d^2} \qquad (51)$$

This stress only occurs at A, where the

beam will ultimately break, and it is evidently possible to take away some of the material between that point and

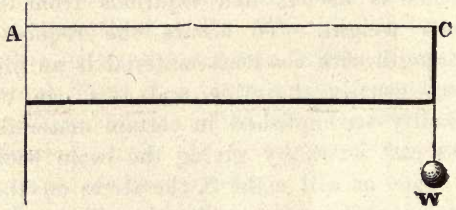


FIG. 47.

C without diminishing the strength. If this be so done that at every point between A and C there shall exist on the outside fibre a stress equal to that at A, the beam will be one of uniform strength, and we shall have attained the greatest economy of material. Let us suppose, the use we have for the beam requires the depth to be uniform. What must be its *plan* in order that S shall be constant in value, or the beam be as liable to break at any other point as at A[5].

In eq. (4) $S=\frac{6\,W\,x}{b\,d^2}$, if we assume S to

be constant, the other side of the equation must be constant also, and since 6 W is constant, and we have made d constant by assuming the depth to be uniform, the whole expression can be constant only when b varies as x.

$\therefore \quad b \propto x$ whence $b = c\,x$ (where $c =$ some constant factor). This equation which is that of a straight line shows that the breadth must vary directly as the length. Hence the *plan* should be a triangle with vertex at C (Fig. 48).

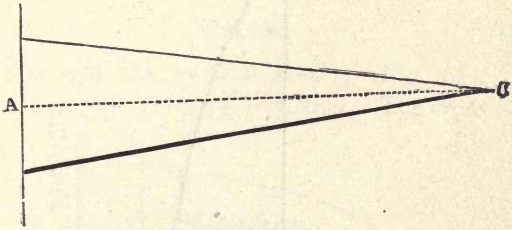


FIG. 48.

On the other hand, if we suppose the breadth to be uniform and wish to have S constant, in the eq. $S = \frac{6\,W x}{b\,d^2}$, x must vary as d, or^2

$$d^2 = c\,x$$

This corresponds to a parabola, and the beam, if the top be straight will have the *elevation* shown in (Fig 49).

Suppose that b varies as d, then $d = n\,b$ (n being a constant) and

$$S = \frac{6\,W\,x}{n^2\,b^3}$$

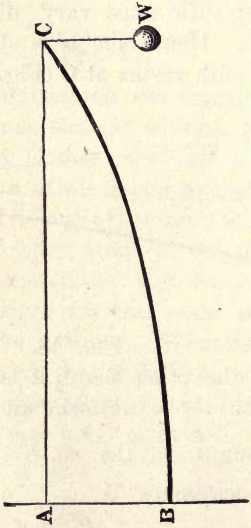


Fig. 49.

To render S constant we must have, $b^3 \propto x$. $\therefore$ $b^3 = cx$ and $d^3 = n^3\ cx$.

These are the equations of a *cubic parabola*. Hence the horizontal section (Fig.

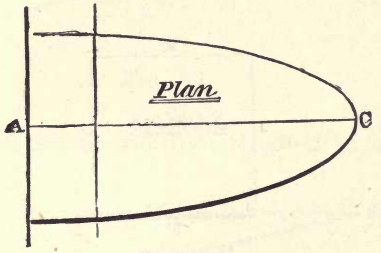


FIG. 50.

50), and the vertical section (Fig. 51), should be curves of that kind. The cross

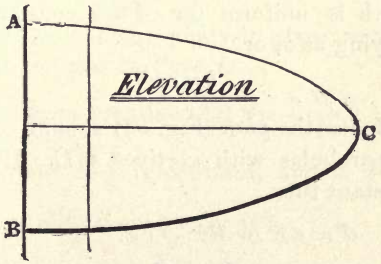


FIG. 51.

section is rectangular as in (Fig. 52).

In *Case II.* we have

$$S=\frac{3\,W x^2}{b\,d^2} \qquad (55)$$

FIG. 52.

Hence, if we make suppositions similar to those above we shall have, when the depth is uniform (or $d=$ a constant) x^2 varying as b, or

$$b=c\,x^2$$

Hence the *plan* (Fig. 53) should consist of parabolas with vertices at C. If b be constant then

$$d^2=c\,x^2 \text{ or } d=\sqrt{c}\,x.$$

This is the equation of a straight line, and

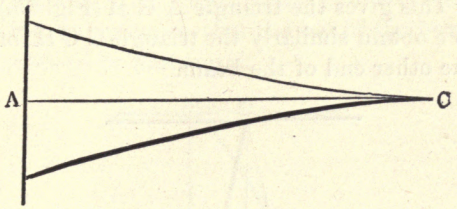


FIG. 53.

gives for the *elevation* the triangle (Fig. 54).

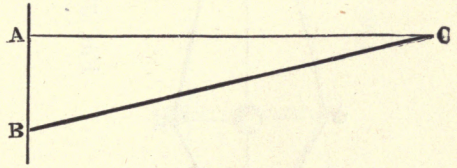


FIG. 54.

In *Case III.* the analysis gives results similar to those in *Case I.*

Thus from equation (21) $S=\frac{6\,W\,n}{l}$.

$\frac{x}{bd^2}$. Here $\frac{6W\,n}{l}$ is constant, and if d be constant also, b must vary as x.

$$\therefore b = c\,x.$$

This gives the triangle A G H (Fig. 55.) We obtain similarly the triangle G C H for the other end of the beam.

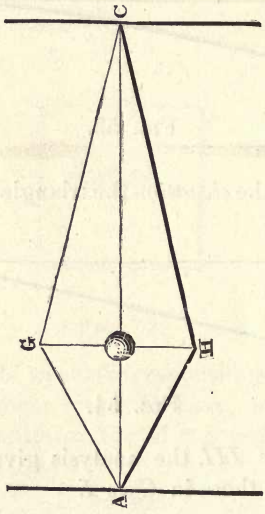


Fig. 55.

If the breadth be constant we have,

$$d^2 = c\,x.$$

which gives a parabola A K (Fig. 56) with vertex at A. So, for the right hand end of the beam the proper elevation is the

parabola C K (Fig. 56). The elevation (Fig. 56) assumes that the top of the beam needs to be horizontal.

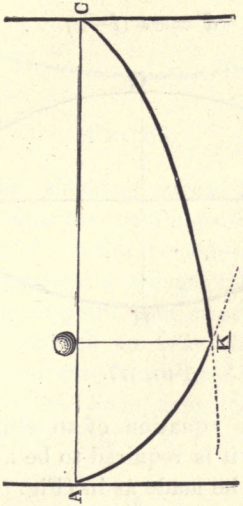


Fig. 56.

In *Case IV.*, equation (37) gives $S=\frac{3w\,x\,(l-x)}{b\,d^2}$: Here, if d be constant in order to render S constant we have

$$b = c\,x\,(l-x).$$

This may be represented by parabolas with vertices at G and H (Fig. 57) opposite the middle of the beam. If b is constant, then,

$$d^2 = c\,x\,(l - x)$$

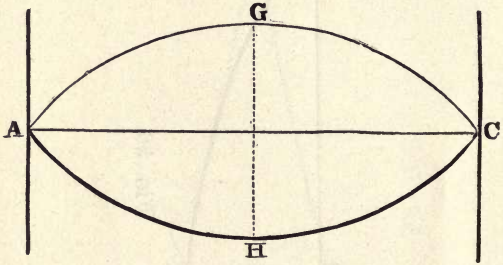


Fig. 57.

which is the equation of an ellipse, and the beam (if it is required to be horizontal on top) may be made as in (Fig. 58).

In these cases of beams of uniform strength, we have so far only considered the *moments* of the weights or the *bending moments* as they are called. But the results are to be modified by the transverse shearing stress. In ordinary rectangular

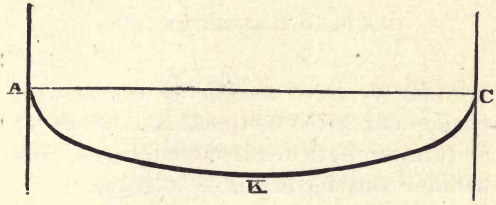


FIG. 58.

beams this shearing stress is so small compared with the bending moment, that it may be left out of consideration. But in beams of uniform strength the ends must not taper to a point, but must always be left large enough to bear the shearing stress. In the case represented in (Fig. 57) the beam should have, near the ends, the shape shown in (Fig. 59).

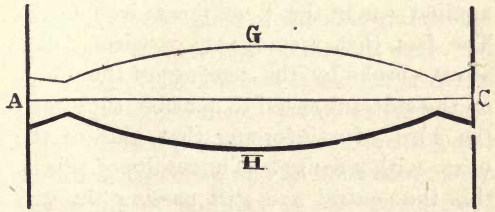


FIG. 59.

DOUBLED FLANGED BEAMS.

So far we have considered beams with rectangular cross sections, and beams of uniform strength deduced from these. We will not consider beams of ⌶ shape.

It is evident from the investigation already given of the condition of stress, in transversely loaded beams, that those portions of the beam nearest the centre bear but a small proportion of the stress, while the contrary is the case with the outside fibres. Hence we would gain strength by moving a considerable portion of that about the neutral axis and placing it on the top and bottom.

The first form in which the idea was applied was in the ⊤ or ⊥ cast iron beam. The fact that rectangular cast-iron beam always broke by the tearing of the fibres on the side subjected to tension, suggested the idea of reinforcing that side of the beam with a flange. The results of this is, that the neutral axis still passing through the centre of gravity of the cross section,

the extreme fibres subjected to compression are farther off than those subjected to tension, and consequently are strained more nearly to their full strength before fracture. This form of beam gives a large increase of strength for the same amount of iron.

It was still plain that the fibres in that part of the web about the neutral axis were but little strained as compared with the fibres on the outside, and it was proposed to leave as little material there as possible, and to place the mass of it in two flanges (I), one above and the other below, giving to these flanges sizes inversely proportional to the tensile and compressive strength of the material. The question then was, how much of the material should be left in the *web*, for plainly all could not be taken. The amount to be left is determined by experiment. If the web is left too thin, the beam will twist and break under the shearing force, and in some cases, from the want of stiffness in the compressed flange.

To simplify the calculations, the web is considered as bearing all the shearing stress,

and no other, and the flanges as bearing all the extension and compression due to the bending moment; and these parts should be proportioned accordingly with due reference to the practical difficulties that sometimes occur. The ordinary formulas for the strength of such beams are gotten by the following approximation: We first neglect the compressive and tensile forces of the web, which are small compared with those of the flanges, and consider it as bearing only the shearing stress. Then as the depth of the flanges is generally small as compared with the depth of the beam, we consider all the fibres in each flange as strained alike, and as bearing the average stress that is brought on that flange. (Fig. 60.)

The resultant of the force on each flange, then, is equal to the stress on a unit of surface (S) multiplied by the flange area (A): that is=S A.

The point of application of the force will be at the middle of the depth of the flanges

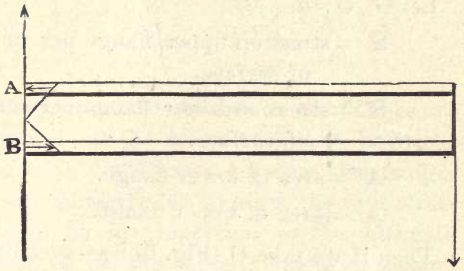


FIG. 60.

(at O and O′, Fig. 61). Fig. 61 shows the

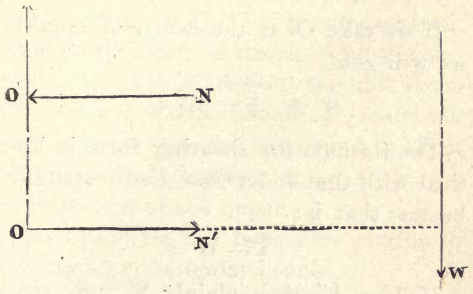


FIG. 61.

forces we have to deal with in the *Case* corresponding to *Case I.* under rectangular beams.

Let O′ O=d.

S′ =stress on upper flange per unit of surface.

S″=stress on lower flange per unit of surface.

A″=area of lower flange.

A′ =area of upper flange.

Then if we take O (Fig. 61) as a centre of moments we have:

$$-N\,d - N'.0.+W\,x = 0 \quad (\text{But } N=S'\,A')$$

$$\therefore S'\,A'\,d = W\,x \qquad (57)$$

If we take O′ as the centre of moments we will get

$$S''\,A''\,d = W\,x \qquad (58)$$

The formula for shearing force is identical with that under *Case I.* of rectangular beams; that is:

$$T = W\,x \qquad (59)$$

If A′ = A″, then plainly S′ = S″ (from equations 57 and 58), or, the forces of tension and compression are equal (as in rectangular beams); but if A′ and A″ are not equal, we have:

$$S' : S'' :: \frac{W\,x}{A'\,d} : \frac{W\,x}{A''\,d} :: A'' : A$$

That is, the unit stresses in the flanges are inversely as the areas. Now, to have the material distributed between the flanges most efficiently for strength the unit stress should be in proportion to the ultimate strength of the material against tension and compression, and hence the areas of the cross sections of the flanges should be *inversely* as the ultimate strength.

Thus, if A D (Fig. 62) be of cast-iron, which is six times as strong against compression as against tension, the unit stress in the lower flange should be made six times as great as in the upper, and to effect this the area of the lower flange should be one-sixth that of the upper.

The *Cases* under ⊥ beams are similar to those under rectangular beams.

Case I.—Beams fixed at one end and loaded at the other.

$$S'\,A'\,d = W\,x, \text{ and } S''\,A''\,d = W\,x \quad (60)$$

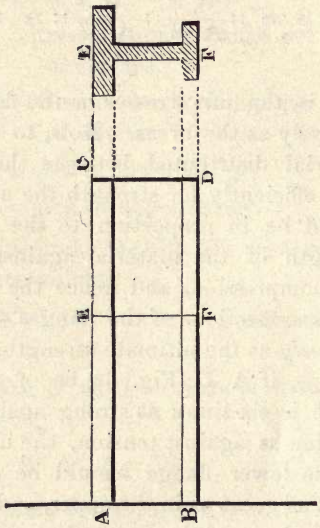


FIG. 62.

Case II.—Beams fixed at one end and loaded uniformly.

$$S' A' d = {}^1/_2 w x^2, \text{ and } S'' A'' d = {}^1/_2 w x^2 \quad (61)$$

Case III.—Beams supported at both ends and loaded at some intermediate point.

(62)

$$W.\frac{n}{l}.x - W(x - m)\ 6'\ A'\ d, \text{ or, } = S''\ A''\ d$$

Case IV.—Beams supparted at both ends and loaded uniformly.

$$S'\ A'\ d = {}^1/_2\ w\ x(l - x) = S''\ A''\ d \quad (63)$$

Case V.—A single moving load over a beam supported at both ends.

$$S'\ A'\ d = \frac{W\ x}{l}\ (l - x) = S''\ A''\ d \quad (64$$

Case VI.—A distributed moving load may be considered as included in *Case IV.*

The formulæ for shearing stress are identical with those in rectangular beams.

The principles of the *uniform strength of beams* may be applied to flanged beams as they were to rectangular beams. The discussion is analogous to that already given.

MOMENT OF RESISTANCE OF BEAMS DETERMINED GEOMETRICALLY.

The following method of obtaining the moment of resistance of beams is of easy application, and in many cases of unsymmetrical cross section is the simplest that can be used:

1. For illustration, take a beam of rectangular cross section. Let G P (Fig. 63) be the cross section at some point of this beam. The stresses on the fibres, as we have already seen, increase just in proportion as we go from the neutral axis towards the upper or lower surface of the beam, and may for any vertical slice (as that at E F) be represented by the ordinates of two triangles, as shown in Fig. 64, where E I (=F J) represents the stress on the outside fibre. For the cross section G P (Fig. 63) the stresses will be represented by two wedges, the bases of which are G M and M R, and the elevations of which are the triangles shown in Fig. 64. The volumes of these wedges give the amount

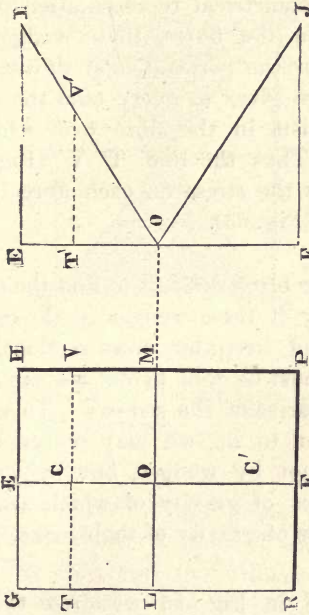


FIG. 64.

FIG. 63.

of compressive and tensile force exerted at the cross section in question, and the points in G P under the centre of gravity of the wedges give the "centres of resistance," or the points of application of the resultants of these forces.

As a geometrical representation of the stresses on the fibres, these wedges are perfect, for the perpendicular ordinate of the wedge gives in every case the stress which exists in the fibre over which it stands. Thus the line T′ V′ (Fig. 64) represents the stress on each fibre in the row T V (Fig. 63).

But it is often difficult to find the centre of gravity of these wedges in the case of curved and irregular cross sections, and yet this must be done before we can know the lever-arms of the stresses. To render this easier to do we may represent the stresses, not by wedges, but by *prisms*, the centres of gravity of which are over the centres of gravity of their bases.

Thus, if in Fig. 65 we draw the two shaded tringles, and conceive prisms of a height = E I (the stress on the outside fibre, Fig. 64), to be constructed on them as bases, we shall have a geometrical representation of the stress on the section G P, less perfect in some respects

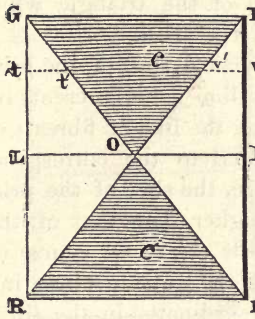


FIG. 65.

than that given by the wedges but better suited to our purpose.

For, note that,

1. The volume of the prism G O H (Fig. 65) is equal to that of the wedge G M (Fig. 63), and the volume of any part of the prism cut off by a plane parallel to the neutral axis, as that whose base t' O v' is equal in volume to the corresponding part of the wedge T M, or since the height of the prism is constant, the stress on the surface G M as we go out from the neutral axis varies

as the area of the triangle which forms the base of the prism.

2. The vertical slice of the prism standing on any line $t' v'$ represents in amount the stress on the line of fibres $t v$, for this stress is equal to the corresponding one in the wedge, the slice of the prism being as much higher than that of the wedge as $t v$ exceeds $t' v'$. Of course (except in the case of the outside fibres in the row G H) each ordinate in the slice of the prism no longer represents the stress on the fibre over which it stands, as was the case in the wedge.

3. The moment of the tensile forces, for instance, will equal the area of the prism G O H multiplied by its height, (E I = stress on outside fibre = S). The centre of resistance of these forces, or the centre of gravity of the prism is at C (Fig. 65), the centre of gravity of the base G O H. The triangle G O H is sometimes called the "effective area" of the surface G M, because a uniform stress on it of an intensity = the unit stress at G H gives the same amount of resist-

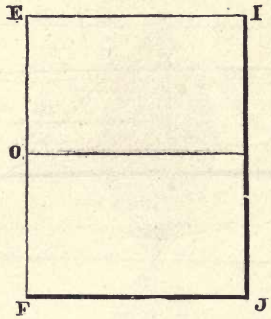


FIG. 66.

ance, as that on the whole area G M, acted on as the latter is by a varying stress.

Considering the stresses represented by the two prisms whose bases are G O H and R O P (Fig. 65), as concentrated at the centres of gravity C and C′ (Fig. 67) of these bases, and taking one of these points as C′, (Fig. 67) as the centre of moments, we have in the case represented in the figure:

(Vol. of prism G O H)$\times$C C′ $=$W x

or if $b =$ breadth and $d =$ depth of beam

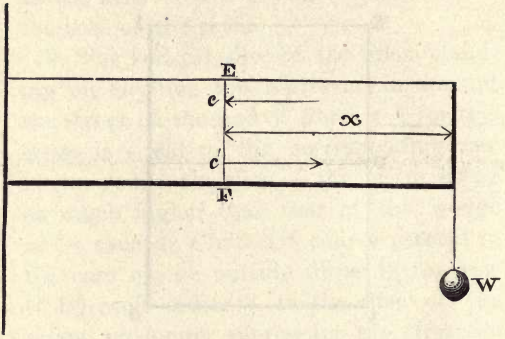


FIG. 67.

$$S(\tfrac{1}{4} b d).\ \tfrac{2}{3}\ d = \frac{1}{6} S\ b\ d^2 = M = W\ x \quad (65)$$

as before.

Corollary. If the beam be square, $b = d$, and

$$M = \frac{1}{6} S\ b^3 \quad (66)$$

II. As a second example, take a square beam so placed that its diagonal will be vertical. Fig. 68 is the cross section. Here we find the base of the prism of stress by points. To find the line in the base of the prism corresponding to the

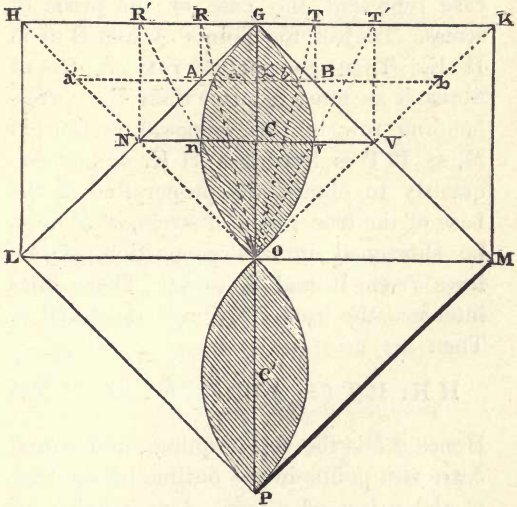


FIG. 68.

stress in any row of fibres, such as A B whose distance from the neutral axis is O X, proceed as follows:

We see that if the cross section were the square of which H L M K is the half, then a' b' would be the line required, since this is the breadth at that point of the triangle H O K, which would in that

case represent the base of the prism of stress. Project the points A and B upon H K. Then the actual row (A B) of fibres is as much shorter than the corresponding row in the supposed section H M, as R T is less than H K, and consequently to obtain the proper line in the base of the true prism of stress, $a'\ b'$ must be shortened in this proportion. Draw lines from R and T to O. These lines intersect the row of fibres at a and b. Then

$$\text{H K: R T } (= \text{A B}) :: a'\ b' :\ ab \qquad (67)$$

Hence $a\ b$ is the line required, and a and b are two points in the outline of the base of the prism of stress. Any number of lines as $n\ v$, &c., may be gotten similarly and the curve drawn through the points $a..n..b..v$, &c., will give the form of the base of the prism of stress. This base is shaded in the diagram.

For any ordinate of the curve O n G, as a X, we have

$$\text{O X: } a\text{ X:: O G: R G} = \text{A X}$$

But $A X = X G$ and making $G O = \frac{d'}{2}$ and putting $O X = x$ and $a X = y$, we have

$$x : y : : \frac{d'}{2} : \left(\frac{d'}{2} - x\right)$$

$$\therefore y = \frac{2}{d'} \cdot \left\{\left(\frac{d'}{2} - x\right)x\right\} \qquad (68)$$

This is the equation of a parabola with vertex at n, half way between H K and the neutral axis. Hence the base of each prism is composed of parts of two symmetrical parabolas.

Areas of the bases. Since the area of a parabola is *two-thirds* of the circumscribing rectangle, the area of each base

$$= {}^2/_3 (G O \times n v)$$

But $n v = {}^1/_2 R' T'$ and $R' T' = {}^1/_2 H K$

$$\therefore n v = {}^1/_4 H K = {}^1/_4 d'$$

$$\therefore \text{Area} = {}^2/_3 \cdot \frac{d'}{2} \cdot \frac{d'}{4} = \frac{1}{12} d'^2$$

The centres of gravity of these bases (and consequently of the prisms) are at C and C′, and the distance

$$C C' = {}^1/_2 d'$$

Hence the moment of resistance of the fibres about C or C′ is

$$M=\frac{1}{24}Sd'^3 \qquad (69)$$

(S=height of prism or stress on external fibres at G and P.)

Corollary. To compare the resistance of the beam in this position with its resistance when lying flat:

Let d=side of the square as L G.

Then $d'=d\sqrt{2}$ and eq. (69) becomes

$$M=\frac{1}{12}Sd^3\sqrt{2}=\frac{1}{6\sqrt{2}}Sd^3 \qquad (70)$$

Comparing this with eq. (66), we see that the beam offers greater resistance when flat in the proportion of

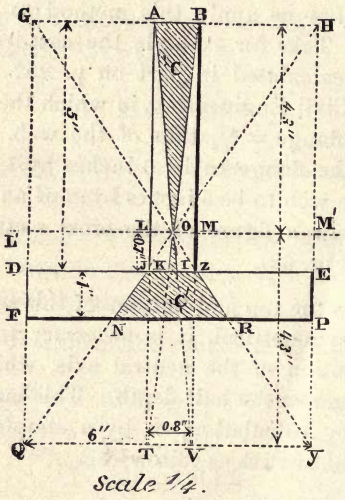
$$\frac{1}{6}:\frac{1}{6\sqrt{2}}$$

In solving problems with diagonally placed beams, place the above value of M equal to the moment of the weight as before.

III. Let us apply this method to a ⊤ beam. Take for example the cast-iron ⊥ beam, calculated in part on p. 257 Rankine's Civil Engineering, in which the area of the flange $= {}^2/_3$ that of the web. Assume the flange to be 6 inches by 1 inch, and the web to be 5 inches by .8 of an inch, and draw a figure of the cross section to scale (Fig. 69).

1. As the top and bottom of this section is not symmetrical, it is necessary to find the position of the neutral axis, which is no longer at the half-depth. This may be done by calculation, or by a simple mechanical process as follows:

The centre of gravity of the cross section, since this last is symmetrical with regard to the vertical line through the middle of the web, must lie on this line. Cut accurately the figure of the cross section out of card board, or tin, or good paper, and suspend it freely by one end of the flange, suspending also from the same point a plummet. Mark the line of the plummet on the card board, and the centre of grav-



Scale 1/4.

FIG. 69.

ity being on this line and also on the middle line of the web, will be at their intersection O. By measurement this point was found to be distant from A B four and three-tenths inches (4.3″), which is also the value by calculations. The line L M drawn through this point is the neutral axis.

2. To determine the bases of the prisms of stress. On the upper side the base is the triangle O A B, if the altitude be taken equal to the stress on the fibres along A B. For if L′ M′ H G were the upper half section, O G H would be the base of the prism and O A B is less than O G H, in the same proportion that L M A B, the real half section, is less than L′ M′ H G. Hence (if the upper be the compressed side) the total compressive force is equal to the prism erected on A O B, with the height equal to the unit stress at A B.

Below the axis L M.—For convenience we should have the height of the tension prism equal to that of the compression one, and the base must be determined under this condition. Complete the large rectangle G H Q Y, making the distance of Q Y below O=4.3 inches. Draw O Q and O Y and the shaded trapezoid J N R Z cut out on the flange by these lines, will evidently be the portion of the base due to the flange. Having prolonged the lines of the web to T and V, draw O T and O V,

and then the shaded triangle O K I will be that part of the base due to the portion (L Z) of the web below the neutral axis.

The total tensile and compressive forces being always equal, and the height of the prisms having been assumed, each $=S'=$ stress on fibres at the distance of A B from O, the bases of these prisms must be equal also. This necessary equality between the area of O A B and that of O K I + J N R Z, affords a means of testing the accuracy of our work in finding the position of O.

3. Area of base O A B. This is,

$= {}^1/_2(\text{L A} \times \text{A B}) = {}^1/_2(4.3 \times .8) = 1.72$ sq. inches.

4. To determine the distance C C′ (Fig. 69) between the centres of gravity of the prisms, which distance is the lever-arm to to be used when one of these points is taken as the centre of moments. These centres of gravity (C and C′) can be readily determined by means similar to those employed in finding the centre of gravity

of the cross-section itself. Thus, cut the shaded areas (Fig. 69) out of card board of paper, and suspending each of them from two points in sucession, draw vertical lines through the points of suspension. The intersection of these two lines gives the centre of gravity. In the present case they may be so simply obtained by calculation that we adopt that method. The centre of gravity of A O B is $=\frac{2}{3}$ the distance from O to A B or O C $=\frac{2}{3}$ (4.3)$=$ 2.87 inches. As to the shaded part below O, by using the ordinary formula for the centre of gravity and taking moments around O, we find the distance

$$O\,C' = \left\{\frac{O\,N\,R \times \frac{2}{3}(1.7) - (O\,J\,K + O\,I\,Z)\frac{2}{3}(.7)}{O\,N\,R - 2\,(O\,J\,K)}\right\}$$

$$= \frac{2.2764 - .01388}{2.0145 - .2975} = 1.318 \text{ inch.}$$

Hence the distance

$$CC' = O\,C + O\,C' = 2.87 + 1.318 = 4.18 \text{ ins.}$$

Hence, since S′ is the height of the

prisms, the moment of resistance of the fibres is

$$M = 4.18 \times 1.72\, S' = 7.19\, S'$$

If it be desired to have M, not in terms of S', the stress along A B, but of S'' the stress on the lowermost fibres (at F P), we have since the stresses increase directly with the distance from O,

$$S' : S'' :: L'G : L'F :: 4.3'' : 1.7''$$
$$\therefore S' = 2.53.\ S'' \text{ and } M = 18.19.S''$$

IV. As an illustration of the great saving of labor sometimes effected by this process, take the steel rail now widely used in England, the cross-section of which is given to scale in (Fig. 70). The determination of the moment here by calculation would be long and tedious. The dimensions of the cross section are given on the figure.

1. The centre of gravity O of the cross section is found by making a template as in the last case, and suspending it freely by a corner. The vertical through the point

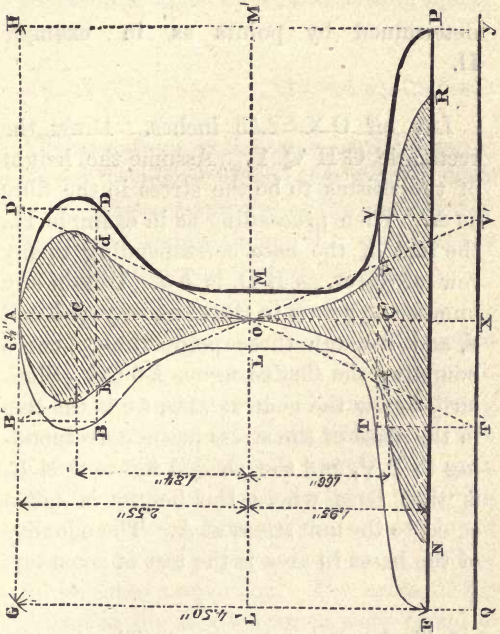


FIG. 70.

of suspension intersects A X at O, which is 2.55 inches below A. Through this point draw L M, the neutral axis.

2. The bases of the prisms of stress are

determined by points as in example II.

Lay off O X=2.55 inches. Draw the rectangle G H Q Y. Assume the height of the prisms to be the stress in the fibre at A. Then proceeding as in example II., the line of the base corresponding to any row of fibres, as B D, is *b d*. Obtain any number of points in the same way as *b* and *d*, and through these points draw a curve bounding the shaded figure A *b* O*d*. Similarly below the neutral axis, *t v* is the line in the base of the stress prism corresponding to T V, and the shaded figure O N R, is that base where the height is taken equal to the unit stress at A. The equality of the bases in area is the test of accuracy.

3. To determine these areas. The simplest plan in the present case is first to find the area of the cross section itself. This is done as follows: The rail in question weighed 84 lbs. per yard, and the steel, of which it was made weighed, .277 lb. per cubic inch.

Hence if A=area of cross section in square inches

(36 . A).277=84. ∴ A=8.4 sq. inches.

Now cut out of the same card board, or paper, templates of the two shaded parts in the diagram, and also of the cross section itself, and weigh them. The ratio of the weights will equal that of the areas.

The comparison of weights may be readily made by the means of a suspended wire, which may serve as a temporary balance, the templates to be compared being stuck on the opposite ends, and one or both moved until the wire is evenly balanced. The weights of the templates being invesely as their distances from the point of suspension of the wire, their areas will be in the same proportion. The areas of the prisms in the case before us were found to be equal each to 2.49 square inches.

4. The centres of gravity of these bases are found in the same way as that of the cross section itself. The point C was thus

found to be 1.84 inches above. O and C' to be 1.66 inches below it. Hence the distance

$$C\,C' = 3.5 \text{ inches.}$$

Therefore, finally, if S′=unit stress at A, the moment of resistance is,

$$M = 2.49 \times 3.5 \times S' = 8.715.\,S'$$

If we desire M in terms of S″ (stress at F P, we have

$$S' : S'' :: 1.84 : 1.66$$

$$\therefore S' = 1.11\,S'' \quad \therefore M = 9.67\,S''$$

V. A circular cross section (Fig. 71). Here the neutral axis of course=L M, passing through the centre. Draw the circumscribing rectangle G Y and obtain the points *t*, *b*, *v*, *d*, &c., in the curve bounding the base, as heretofore. Through the points so found draw the curves.

Determine the areas of the bases of the stress prisms by comparing them with the half-square G L M H. Thus, if a template is cut to the surface of one of these bases it will just equal in weight the template cut

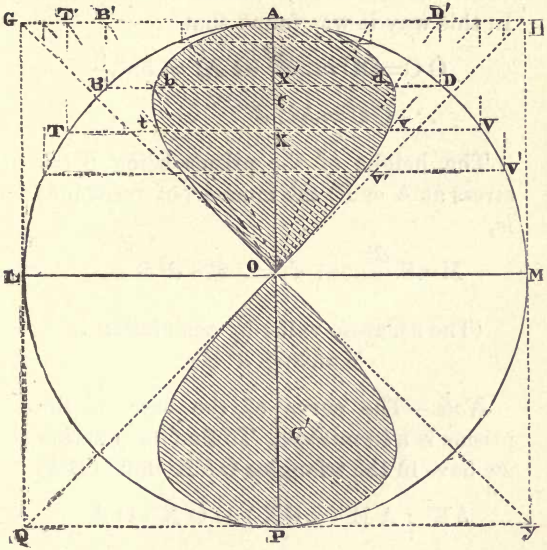


FIG. 71.

to the surface A G L O *t b* A, or, in other words, the shaded surface A *b* O *d* A is just one-third of the rectangle G M, or $\frac{1}{3} \cdot \frac{d^2}{2} = \frac{1}{6} d^2$.

The centres of gravity C and C′ are

found as before by means of templates. In this way it was found that

$$O\,C = OC' = .587\,(O\,A) = .587.\frac{d}{2}$$

$$\therefore C\,C' = .587\,d$$

The height of the prisms being S (= stress at A or F) the moment of resistance is,

$$M = S.\frac{d^2}{6}(.587\,d) = .0978\,d^3\,S$$

(The accurate value by calculation is

$$M = .0982\,d^3\,S.)$$

Note.—The curve of the base of the prisms is a *lemniscate.* To find its equation we have in the triangles O b X′ and O B′A,

$$b\,X' : A\,B'\,(=B\,X') :: O\,X' : O\,A$$

And taking the vertical axis as that of X, and the horizontal one, as that of Y, the origin being at O, and calling the co-ordinates of the circle x' and y', and those of the *lemniscate* x and y we have:

$$y : y' :: x : R, \text{ or } y : \sqrt{R^2 - x^2} :: x : R$$

$$\therefore R^2\,y^2 = x\,(R^2 - x^2).$$

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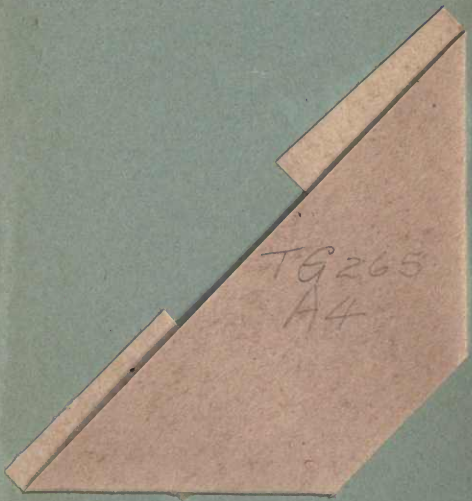

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